

## ESSI COLPRO

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### ABSTRACT

This paper presents ESSI COLPRO, an analytic simulation model that enables rapid tradeoff studies and optimization on COLPRO systems. The simulation is an end to end model of a COLPRO system, which includes the major indirect contributors to weight and power of thermal load and flow. The thermal load models include all major contributing factors to load, and utilizing the material properties of the COLPRO system derives the thermal loads and the power requirements of the shelter. By optimization, one can reduce the peak and average thermal needs, reduce overall weight of the system, reduce the overall requirements and fuel consumption needs. The latter can significantly reduce the fuel logistics tail requirements of COLPRO systems. All this is complementary to the transformation needs of the Army, but also meets the needs of the Navy, Marines and Air Force. The model will be explained together with specific examples of optimization and logistic fuel tail reductions.

### INTRODUCTION

Collective protection of existing buildings, new buildings, existing tents and new tents hinges on the flow of clean filtered air and of the isolation, nominally by overpressure, from outside contamination. Whereas, the use of liners or various tent materials represents an acquisition cost, the constant flow of replenished clean air represents a huge proportion of the maintenance and running cost of the system. With the added requirement that the clean air be held at some nominal comfort temperature for personnel, the HVAC load (and the associated electrical power to drive that HVAC system) become the critical operating cost to run the system. With such subsidiaries as EASI and KECO involved in collective protection and chem/bio hardened HVAC systems, ESSI (the parent company) decided to build an analytic model that could predict the major load requirements from a thermal load perspective for any of the above collective protection scenarios. Such a model could then be exercised parametrically to tradeoff various contributing factors such as thermal insulation versus weight and cost, versus operating costs, thereby yielding some optimal criteria for selecting a given solution for a collective protection system.

In the most simplistic sense, the size, physical attributes, physical material properties, location and time of year, and the collective protection requirements for air flow dictate the thermal load. By parameterising the weight per BTU/hr and the electrical power per BTU, as well as the weight per kwatt of electrical power and its efficiency, the overall weight of the equipment can be established, but moreover the weight of the fuel consumed over a given period can be established.

Described below is the model in parametric detail. As an example of optimization, we used the program to predict and optimize loads for the proposed JTCOPS tents. Although we optimized for all the tents considered under JTCOPS, we selected the larger MSS tent as the example used here for illustration.

## COMPONENTS OF HEAT LOAD

In modeling the thermal load, we have assumed six heat sources. These components of the heat load are:

1. Solar radiation
2. Conduction from the higher temperature surroundings
3. Personnel in the controlled space
4. Active equipment in the controlled space
5. Latent heat of condensation of humidity at the cooling coils
6. Make up air

The latter four components are essentially independent, whereas the first two are dynamically interactive and need to be treated as such.

## SOLAR RADIATION

For earth bound systems, solar radiation is a very predictable thermal load. The position of the building or tent on the earth's surface together with its orientation, and its surface area normal to the sun's warming rays at that particular time of day, month or year are readily predictable.

All that is required is to establish the solar altitude, air mass, the reflection and absorption properties of the building surface, and the building's orientation and its projected area.

### Solar Altitude

The solar altitude, or position above the horizon, affects the influx of solar radiation in two ways. First it determines the projected area of the tent outer surface perpendicular to the sun's rays, which is an important variable because if the angle is more oblique, a smaller amount of power strikes the surface and the solar load is less. Secondly, when the sun is lower in the sky less power per unit area reaches the Earth's surface because of the longer path through the attenuating atmosphere. We shall consider both of these effects in later sections, but here we treat the problem of solar position.

We have used an astronomical model whose efficacy was prior proven. It is based on algorithms from a useful little book (**ref 1**) that actually predates the personal computer era. The only inputs required are latitude, longitude, date, and time.

First the date, including day, month, and year, is used to fix the position of the sun on the celestial sphere; that is, the solar position against the background of fixed stars. This celestial position is specified by declination (angle above the celestial equator) and right ascension (angle around the celestial equator, starting from the vernal equinox). The position is nearly the same for the same date every year, except for corrections that need to be made for leap years.

Then, to accommodate the rotation of the Earth on its axis, the celestial sphere (and all the celestial bodies fixed upon it) is assumed to rotate east to west in diurnal motion. The values of latitude and longitude for the observer and time of day allow the calculation of the solar position, or any other

point on the celestial sphere, with respect to the observer's horizon. This location is specified by altitude (angle above the horizon) and azimuth (angle around the horizon starting from the north point). For our purposes it is the altitude which is important. We have used a time of day input in Universal Time (UT), formerly known as Greenwich Mean Time (GMT). If the time zone of the observer is known then it is easy to translate UT to local time, but it is difficult to automate the translation of observer latitude and longitude into the local time zone because of the irregularity of time zone boundaries around the world to accommodate political realities.

#### Solar Radiation Absorbed per Unit Area

Rather than carry out a full spectral analysis we used a spectral band approach. One reason is that the main reference (ref 2) on solar radiation already has the flux averaged over the individual bands. Table 16-11 from this reference is given in part below

Irradiance Normal to the Sun's Rays at Sea Level in W/m <sup>2</sup>					
	Air Mass =1	Air Mass =2	Air Mass =3	Air Mass =4	Air Mass =5
.29-.40 $\mu\text{m}$	40.1	19.8	10.0	5.4	2.7
.40-.70 $\mu\text{m}$	419.7	327.8	258.6	205.8	163.7
.70-1.1 $\mu\text{m}$	309.2	267.5	233.4	205.1	181.5
1.1-1.5 $\mu\text{m}$	95.3	70.7	57.0	48.1	40.7
1.5-1.9 $\mu\text{m}$	50.8	45.1	41.0	38.0	35.2
1.9- $\infty$ $\mu\text{m}$	12.8	9.2	7.5	6.5	5.8

The first band is in the ultraviolet region of the spectrum. The second band corresponds to the visible spectrum, or light. The third, fourth, and fifth bands correspond to atmospheric transmission bands in the near infrared. The sixth band contains all the rest of the sun's energy. For simplicity we used only the middle four bands for our calculations, noting that the ultraviolet and infrared energy past 1.9  $\mu\text{m}$  represents only a few percent of the total, decreasing to negligible at higher air masses.

Air mass is defined as the ratio of the path length of radiation through the atmosphere at any given angle to the path length through the atmosphere toward the zenith. As such, the air mass is approximately equal to the secant of the zenith angle,  $z$ , where zenith angle is the complement to altitude. The reason that the relationship is only approximate is that atmospheric refraction affects the actual path length at larger zenith angles (sun lower in the sky). For our purposes, we ignore the effects of refraction.

$$m = \sec(z) \quad (1)$$

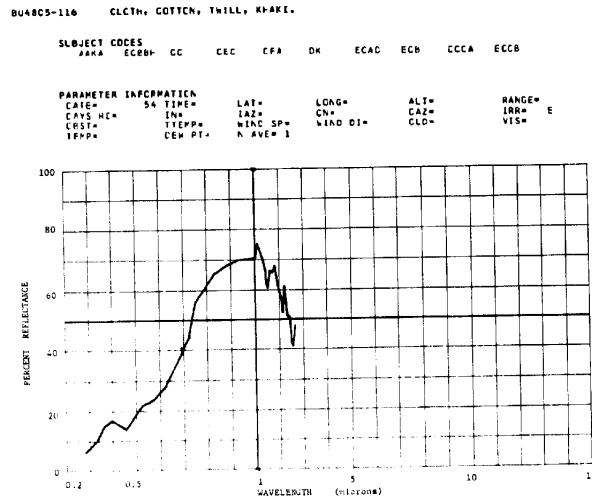
$$z \equiv 90^\circ - alt \quad (2)$$

In order to map out a smoothly varying profile of solar radiation versus time of day, we used a least-squares curve-fitting routine, along with the table values, to generate a function specifying irradiance versus  $m$  for each of the four pertinent spectral bands. It turned out that for all four bands of interest, the solar irradiance fit very well to the general form

$$E_s = a + b \cdot \exp\left(-\frac{m}{c}\right) \quad (3)$$

with different values for  $a$ ,  $b$ , and  $c$  in the different bands. Then, as the altitude is calculated for various times of day, equations (1), (2), and (3) allow calculation of the corresponding solar irradiances.

In each band, part of the solar irradiance is reflected at the outer surface of the building or tent, and the remainder absorbed or transmitted. As noted before, we consider the solar heat load to consist of all the radiation that is not reflected at the outer surface. To calculate this quantity, we must have an average reflectance for each spectral band. For our test example, a large data compilation (ref 3) for the USAF provides the necessary information for a number of materials. We assumed khaki cotton twill to have the closest optical characteristics to the tent outer surfaces. Its spectral reflectance from reference 3 is shown below.



As can be seen, the reflectance varies considerably in the visible region, 0.4-0.7  $\mu\text{m}$ , but we settled on an approximate average of 22%. In the infrared bands the variability is less and we estimated 65% reflectance for each of the other three bands. The fraction converted to heat in the controlled space is then 78%, 35%, 35%, and 35% respectively for each of the bands. Therefore, once the four band values for solar irradiance are calculated for a given solar altitude by equation (3), the total absorbed irradiance is found by summing the four irradiance values multiplied by their respective percentage values. A conversion factor of 3.414 is required to convert watts to BTU/hr.

Because irradiance is power per unit area, at this point we will have total solar heat load absorbed per unit projected area of the building or tent in question. By projected area, we mean area perpendicular to the direction of the sun.

### Projected Area

The actual projected area of each building or tent is a complicated function of the solar altitude and azimuth, building/tent geometry, and building/tent orientation on the Earth's surface. For our chosen example here, we made the simplifying assumption that the tent always had its long axis perpendicular to the sun's direction, a kind of worst case scenario. With this assumption, solar azimuth becomes unimportant, and only the solar altitude determines the projected area. It should be noted that this assumption generally implies that the tent turns on the Earth's surface as the sun undergoes its diurnal motion. The local noon orientation would be with the long axis east-west.

The MSS is a semi-cylindrical tent and gives the simple result for projected area:

$$A_p = L \cdot R[1 + \cos(z)] \quad (4)$$

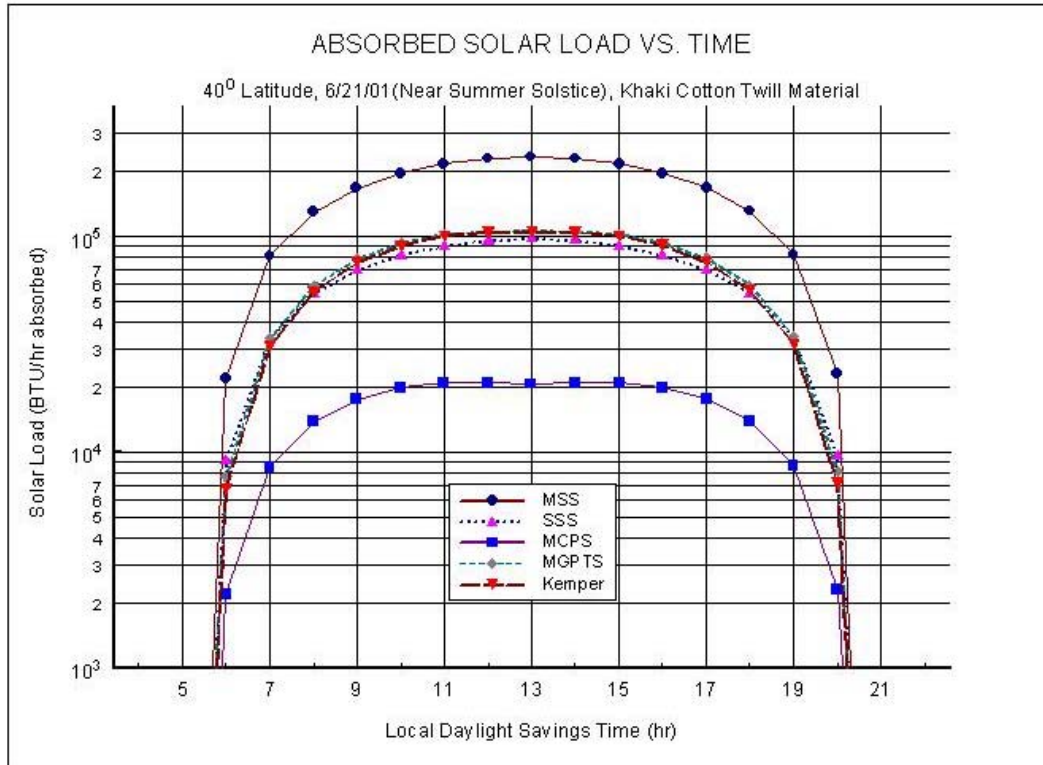
where  $L \equiv$  long axis length

$R \equiv$  outer radius

So for the MSS, projected area in  $\text{m}^2$  is given by  $A_p = 72.46[1 + \cos(z)]$

### Sample Results for MSS Tent

Using the equation above, we may trace out the solar heat load on the MSS for any day at any location. The results shown below are for  $40^\circ$  latitude (about that of St. Louis) and near the summer solstice.



From these results, it can be seen that the absorbed solar load (dependent upon the spectral properties of the outer material) can be very high. In order not to change the signature of buildings or tents, and generally because liners are used, we decided to consider that the model encompass a liner. The (thermal) properties of the liner can be significantly different from that of the outer surface. Indeed one can tailor modern fabrics to have a whole series of properties. One way of reducing this overwhelming solar radiation is to use liners with more reflective properties than the outer surface.

## HEAT LOAD WITH REFLECTIVE INNER LINING

### Introduction

In this case, the assumption that the tent or building outer surface remains at ambient temperature, is no longer viable. If all the radiation from the outer surface is reflected at the inner surface,

then the outer surface will likely rise above ambient and that higher temperature will give rise to greater conductive heat load. In other words, we can no longer use the assumption that the two components, conduction and solar, are independent. Instead we must find the equilibrium temperature of the outer surface before proceeding further with the heat load analysis.

### Equilibrium Temperature

We may find the equilibrium temperature by assuming a dynamic balance between the heat entering and the heat leaving a unit area of the outer surface. To this end, we assume a solar heat input per unit area labeled  $S$ , a conductive heat loss to the inner surface, a radiative heat loss to the surroundings, and a convective heat loss to the atmosphere.

### Solar Heat Input

One problem here is that only part of the building/tent is in direct sunlight at any given time. The solar input on that part is high, while the input on the shaded portion of the building/tent is zero. The lateral heat flow through the fabric of the outer surface we assume to be very low because of the small cross-sectional area. This assumption means that there would really be two equilibrium temperatures for the outer surface, one for the area in the sun and one for the area in the shade. Rather than contend with this complication, we simply found an average input per unit area,  $S$ , by dividing the total absorbed solar radiation, as calculated previously, by the lateral exposed surface area (without ends or floor surface areas). This method should provide a kind of average surface equilibrium temperature, somewhere between the two described above. Because of differences in details of tent construction between the types, the resulting value for  $S$  at any given time of day is slightly different for the different types, but they are all close together so that an average can be used. For example, the maximum value of  $S$  at local noon for 40° latitude near the summer solstice is about 90 BTU/ft<sup>2</sup>.

For the absorbed solar radiation we use all the radiation that is not reflected. Actually there is some small percentage which is neither absorbed nor reflected, but rather transmitted. That portion is 20% or below in every spectral band used, and it would be reflected by the inner lining back to the outer surface, where a small portion would then be transmitted again. That portion that is transmitted twice (and therefore never contributes to the solar load) must be less than 20% of 20% = 4%, and therefore is ignored in our analysis.

### Conductive Heat Loss

Here we assume the inner surface is at some spec temperature,  $T_s$  (29.5°C maximum), and the outer surface is at the equilibrium temperature,  $T_e$ , (to be determined). The law for conductive heat flow per unit area is

$$H_c = \frac{1.8(T_e - T_s)}{R} \quad (5)$$

The factor of 9/5=1.8 is included in order to convert from a temperature difference in kelvins, required by the Stefan-Boltzmann law for radiation, to a temperature difference in °F required by the units of  $R$ , (BTU/hr ft<sup>2</sup> °F)<sup>-1</sup>.

### R-Values

Generally in our analyses, the value of  $R$  has been used as a parameter, ranging from 1 to 5. However, we extend the values farther in this case to allow for the possibility of a dead air gap between the outer and inner surfaces.

The thermal conductivity of air varies with temperature, but is roughly  $65 \times 10^{-6}$  (cal/s cm<sup>2</sup>)(cm/°C) at 110°F. Converting units, we find  $k_{\text{air}} = .189$  (BTU/hr ft<sup>2</sup>)(in/°F). Then  $R_{\text{air}} = t/k_{\text{air}} = 5.30t$

where  $t$  is the thickness of the conductor in inches. It can be seen that a dead air gap of 2 inches can lead to an  $R$  value of 10.

#### Radiative Heat Loss

As noted above, the Stefan-Boltzmann Law governs radiative heat loss to the surroundings (any radiation from the outer surface inward is reflected by the inner reflective lining). In the thermal IR range we are safe in assuming that all objects are approximately blackbodies with emissivities near unity. Therefore radiative heat loss per unit area is given by

$$H_r = \sigma (T_e^4 - T_a^4) \quad (6)$$

where  $\sigma = 1.797 \times 10^{-8} \text{ BTU}/(\text{ft}^2 \text{ hr K}^4)$

$T_a$  = ambient exterior temperature (49°C maximum)

#### Convective Heat Loss

We have found two different empirical formulas for natural convection off surfaces in the atmosphere. The first (ref 4) uses temperature difference along with a convection coefficient which, itself, is a function of temperature difference. For a vertical surface

$$H_{conv} \cong h (T_e - T_a) \quad (7)$$

where  $h \cong .424 \times 10^{-4} (T_e - T_a)^{1/4} \text{ cal}/(\text{s cm}^2 \text{ }^\circ\text{C})$

The second algorithm (ref 5) uses the Langmuir formula. This gives, for a vertical surface,

$$H_{conv} \cong [\Phi(T_e) - \Phi(T_a)]/0.45 \quad (8)$$

where  $\Phi$  is a tabulated function of  $T$

The units of equation (4) are the same as those of equation (3):  $\text{cal}/(\text{s cm}^2)$ . When a curve fit is made to the tabulated function of  $T$  in the range  $T = 100\text{K} - 400\text{K}$ , we find

$$\Phi(T) \cong 0.0002 + 1.7 \times 10^{-7} T^{1.914} \quad (9)$$

with  $T$  in K.

Both sources assert that a horizontal surface would give larger convective losses than those specified above. The first source suggests 25-30% greater, while Langmuir says only 10% greater. Although the two approaches appear quite different, a calculation by each method for the convective loss from a vertical surface at 400K to the atmosphere at 300K, leads to  $0.0134 \text{ cal}/(\text{s cm}^2)$  from the first and  $0.0150 \text{ cal}/(\text{s cm}^2)$  from the second. These answers are in reasonable agreement, considering the empirical nature of the formulas. We chose to use the Langmuir formula, along with the curve fit of equation (5), because the source is about 30 years more recent.

The conversion of units to BTU/(hr ft<sup>2</sup>) in equation (4) leads to a constant 29491 multiplying the  $\Delta\Phi$ . For horizontal surfaces, 10% larger would change the constant to 32440. Since our building/ tent surfaces can be considered as a combination of horizontal and vertical we compromise at a constant of 30000.

$$H_{conv} \cong 30000[\Phi(T_e) - \Phi(T_a)] \quad (10)$$

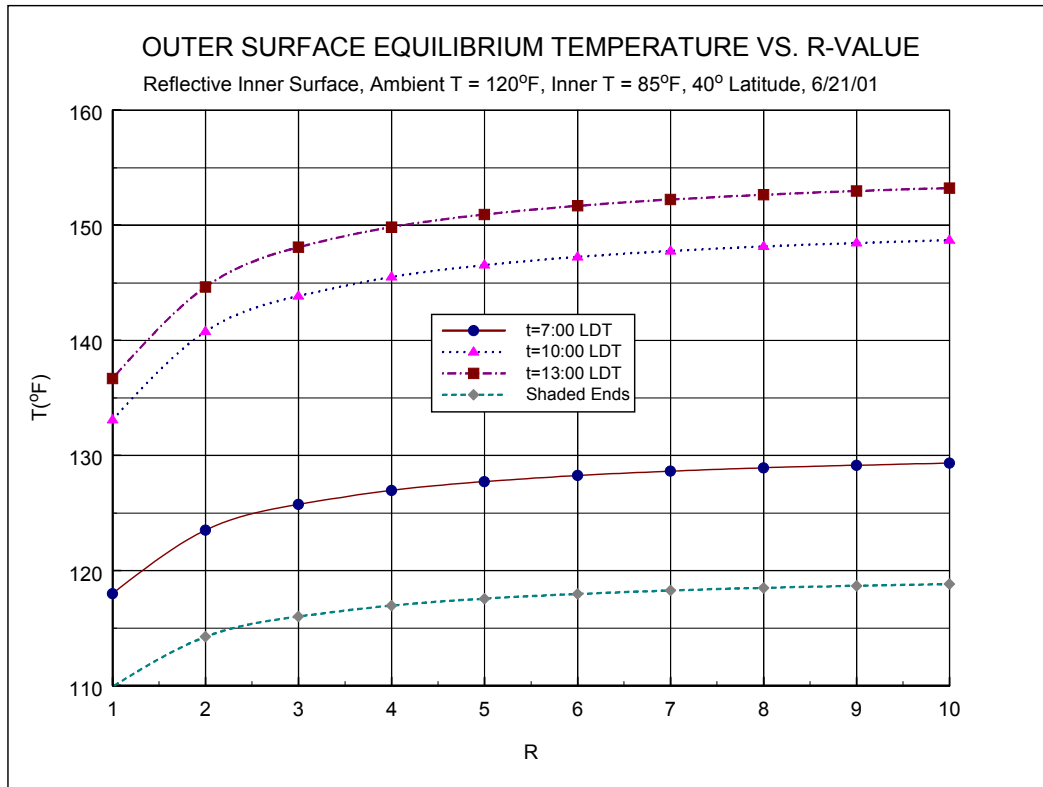
### Equilibrium Equation

The substitution of all the terms into the equilibrium equation, expressing the balance of incoming and outgoing heat per unit area of the outer surface, gives

$$\sigma T_e^4 + \frac{1.8}{R} T_e + 30000\Phi(T_e) - \left[ \sigma T_a^4 + \frac{1.8}{R} T_s + 30000\Phi(T_a) + S \right] = 0 \quad (11)$$

All temperatures in equation (7) can be considered to be in K, since temperature differences in °C are the same as differences in K.

The equilibrium temperature is the root of this equation; that is, the value of  $T_e$  that makes the left side equal zero for given values of  $S$  and  $R$ . The results are shown in the graph below, for the MSS, for several different times of day (different  $S$  values) and for the shaded ends ( $S=0$ ).



Local noon, or 13:00 Daylight Savings Time, represents the highest solar load and, accordingly, the highest surface temperatures. The temperature profile is symmetric about local noon, so the curve for 16:00 LDT should look much the same as that for 10:00 LDT. Note that with low solar input, the outer surface equilibrium temperature can fall below ambient, and with no solar input the equilibrium temperature is below ambient for any R-value.

## HEAT LOAD

Using these outer surface temperatures we may now recalculate the total heat load for any building or tent type. We divide the load up into six components. There are the two fixed components (independent of R-value and exterior conditions): personnel,  $Q_p$ , and electrical equipment,  $Q_e$ . The condensation of humidity at the cooling coils represents another heat load,  $Q_L$ , independent of tent R-values and surface areas, but depending on the exterior and interior conditions. There is the conductive heat load from the lateral surfaces exposed to the sun, labeled  $Q_1$ . There is the conductive heat load,  $Q_2$ , from the ends which will be shaded according to our assumption of the long axis of the tent perpendicular to the vertical plane containing the sun. There is also the conductive heat load through the floor, which receives no solar input and also has no convective losses to the atmosphere.

### Conductive Heat Loads

All three conductive heat loads are calculated from the general equation.

$$Q = \frac{A}{R} (T_2 - T_1) \quad (12)$$

For  $Q_1$  we use the lateral surface area for  $A$  (excluding ends and floor), and the equilibrium temperature as calculated above for  $T_2$ . Therefore  $Q_1$  will be a function of  $R$  and  $S$  (solar position). For  $Q_2$  we use the surface area of the two ends for  $A$ , and the equilibrium temperature for  $S = 0$ .  $Q_2$  will be a function of  $R$  only. For the floor area, we revert to our original assumption that the outer surface is at ambient temperature, while the inner surface is at the spec temperature. This assumption probably overestimates the heat load from the floor. The floor for a tent such as the MSS probably cannot have a dead air insulating space and so we set a fixed R-value of 1 or 2 for this area. Therefore for  $Q_3$  we use the surface area of the floor for  $A$ ,  $T_2$  = ambient external temperature, and  $R$  either equal to 1 or 2. We use the spec internal temperature for the value of  $T_1$ .

### Condensation Heat Load

When water condenses on the cooling coils it gives up a latent heat of condensation directly to the coolant, which becomes part of the heat load for the air conditioner. Such condensation is an isobaric process, and so the heat transferred is equal to the change in enthalpy of the water (ref 6):

$$L = h''' - h'' \quad (13)$$

where  $h'' \equiv$  enthalpy of liquid water at condensation temperature

$h''' \equiv$  enthalpy of water vapor at condensation temperature

Reference 1 contains a table of these values for water at a range of temperatures. For  $50^\circ\text{F} = 10^\circ\text{C}$  the values are  $h''' = 2519 \text{ J/g}$  and  $h'' = 42 \text{ J/g}$ . The latent heat of condensation at this temperature (the assumed temperature at the cooling coils) is therefore  $L = 2477 \text{ J/g} = 2.35 \text{ BTU/g}$ .

Although we have the heat load for each gram of condensed water, we must now consider how many grams of water we have in the air we are considering. An empirical formula from the work on LOWTRAN atmospheric models is alleged (ref 7) to be accurate to 1% in the temperature range -50°C to +50°C. This gives the saturation concentration of water vapor in the atmosphere as a function of temperature.

$$C_s(T) = A(T) \left[ 18.9766 - 14.9595 A(T) - 2.4388 A(T)^2 \right] \quad (14a)$$

with

$$A(T) \equiv \frac{273}{T} \quad (14b)$$

where

$T \equiv$  temperature in kelvins

$C_s =$  saturation concentration in  $\text{g/m}^3$

The relative humidity is defined from

$$RH \equiv \frac{C}{C_s(T)} \quad (15)$$

where

$C \equiv$  actual concentration

If we know the temperature and relative humidity for any quantity of air, we may use equations (10) and (11) to calculate the actual water vapor concentration in  $\text{g/m}^3$ . For example the spec range of interior RH is 30% to 60%. At the maximum spec temperature of 29.5°C, we obtain

$$\begin{aligned} C_1 &= 0.30 \times C_s(29.5+273) = 9.12 \text{ g/m}^3 \\ C_2 &= 0.60 \times C_s(29.5+273) = 18.24 \text{ g/m}^3 \end{aligned}$$

The outside air is at some ambient temperature,  $T_a$ , and some ambient relative humidity,  $RH_a$ , implying some ambient water vapor concentration,  $C_a$ . If this air is introduced into the tent at some flow rate  $F$  (in  $\text{m}^3/\text{hr}$ ), then the ambient concentration must be reduced to at least the value  $C_2$ , calculated above. The concentration of water removed,  $C_r$ , (condensing) must then be

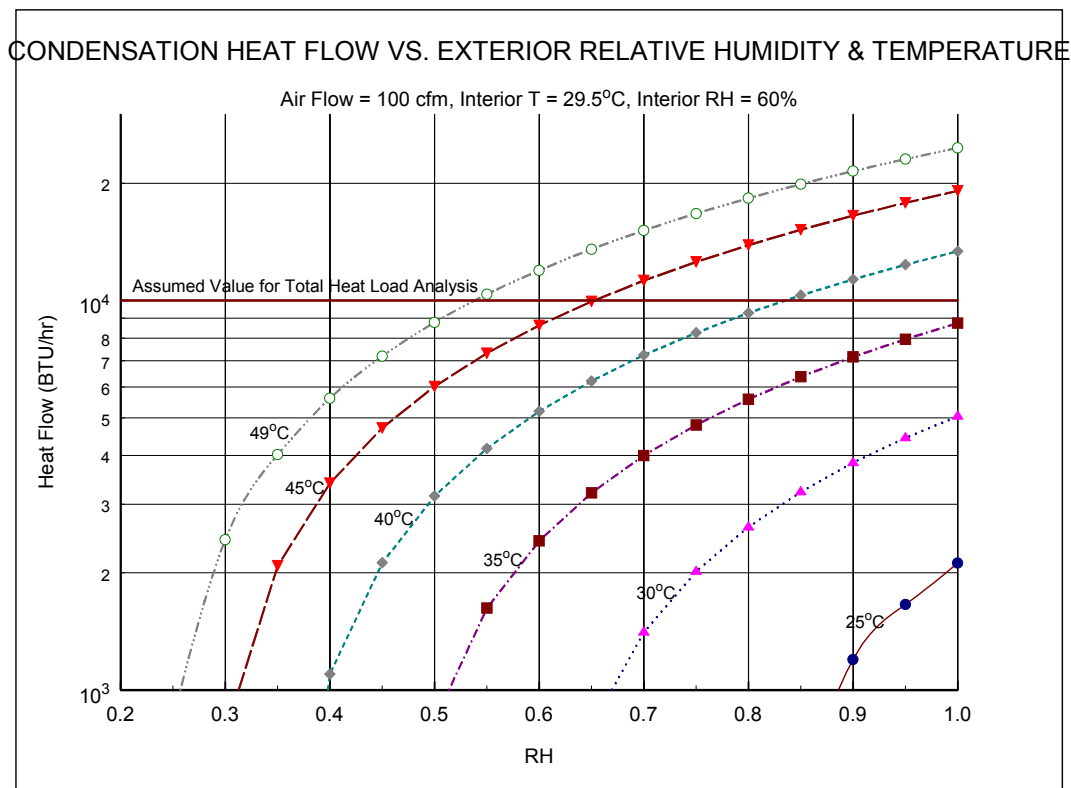
$$C_r = C_a - C_2 \quad (16)$$

The heat load would be the flow rate ( $\text{m}^3/\text{hr}$ ) times the grams of water condensed from each  $\text{m}^3$ ,  $C_r$ , times the latent heat per gram.

$$Q_L = F C_r L \quad (17)$$

For the medium tents  $F = 100 \text{ cfm} = 170 \text{ m}^3/\text{hr}$ , and we have already established the value of  $L$  to be 2.35 BTU/g. The value of  $C_r$  depends on the ambient temperature and relative humidity because the value of  $C_a$  does. Ambient temperatures can range up to 49°C (120°F) and relative humidities up to 100%. However it is extremely unlikely that both maximum values occur together. Such maximum temperatures

are likely only to occur in desert areas, in which case humidities are likely to be 10% or less. We would venture to guess that no such combination of temperature and humidity (49°C and 100% relative humidity) has ever occurred naturally since humans have walked the face of the Earth. The implications of temperature and relative humidity on the condensation heat flow are illustrated by the graph below.



It can be seen that the ambient temperature must be above 35°C = 95°F before the condensation heat load can be as large as 10000 BTU/hr at any relative humidity. Even at the maximum temperature of 49°C, the relative humidity must be 55% to reach this heat load. Therefore, for the medium and small tents we set QL = 10000 BTU/hr; for the large tents, which have twice the airflow, we use QL = 20000 BTU/hr.

#### Personnel in the Collective Protection Environment

For modeling purposes, all that is required is an estimate of the number of people occupying the space. It can be assumed that an active number of the military will have a caloric intake of 3,000 calories/day, from which an estimate of the thermal load per person can be established.

#### Active Equipment in the Collective Protection Environment

The operating equipment will provide both an electrical load and a thermal load. If the ohms electrical load is known, it directly reflects in watts the thermal load.

#### Make Up Air: Outside Air Flow Heat Load

There is one last component of heat load to be considered.. That is the cooling required for the introduction of outside air into the tent. Any quantity of outside air, assumed to be at 49°C, introduced into the tent must be cooled to 29°C.

If we take the flow rate,  $F$  in  $\text{m}^3/\text{hr}$ , times the molar density of the air, in  $\text{moles}/\text{m}^3$ , we shall have the flow rate in  $\text{moles}/\text{hr}$ . Then multiplying by the molar heat capacity of air at constant pressure,  $c_p$ , and the temperature change gives the heat load.

The molar density of air can easily be found from the Ideal Gas Law:

$$\rho_m \equiv \frac{n}{V} = \frac{P}{RT} \quad (18)$$

where  $n \equiv$  number of moles of air  
 $V \equiv$  volume occupied by  $n$  moles  
 $P \equiv$  pressure ( $\text{N}/\text{m}^2$ )  
 $R \equiv$  molar gas constant =  $8.317 \text{ J}/(\text{mole K})$   
 $T \equiv$  air temperature in kelvins

The air flow heat load is therefore given by

$$Q_a = \frac{F P}{R T} c_p \Delta T \quad (19)$$

The molar heat capacity of a gas is actually a weak function of temperature. An empirical formula, good to 2%, is given by (ref8)

$$c_p = a + bT + cT^2 \quad (20)$$

For diatomic nitrogen, which is 80% of the atmosphere,  $a = 6.30 \text{ cal}/(\text{mole K})$ ,  $b = 1.819 \times 10^{-3} \text{ cal}/(\text{mole K}^2)$ , and  $c = -.0345 \times 10^{-6} \text{ cal}/(\text{mole K}^3)$ . These values imply that at  $49^\circ\text{C} = 322\text{K}$

$$c_p = 6.85 \text{ cal}/(\text{mole K})$$

For diatomic oxygen, the other major constituent of the atmosphere, the coefficients are slightly different, but the final answer is  $c_p = 7.06 \text{ cal}/(\text{mole K})$ . Therefore an approximate value of  $6.9 \text{ cal}/(\text{mole K})$  should suffice for our calculations.

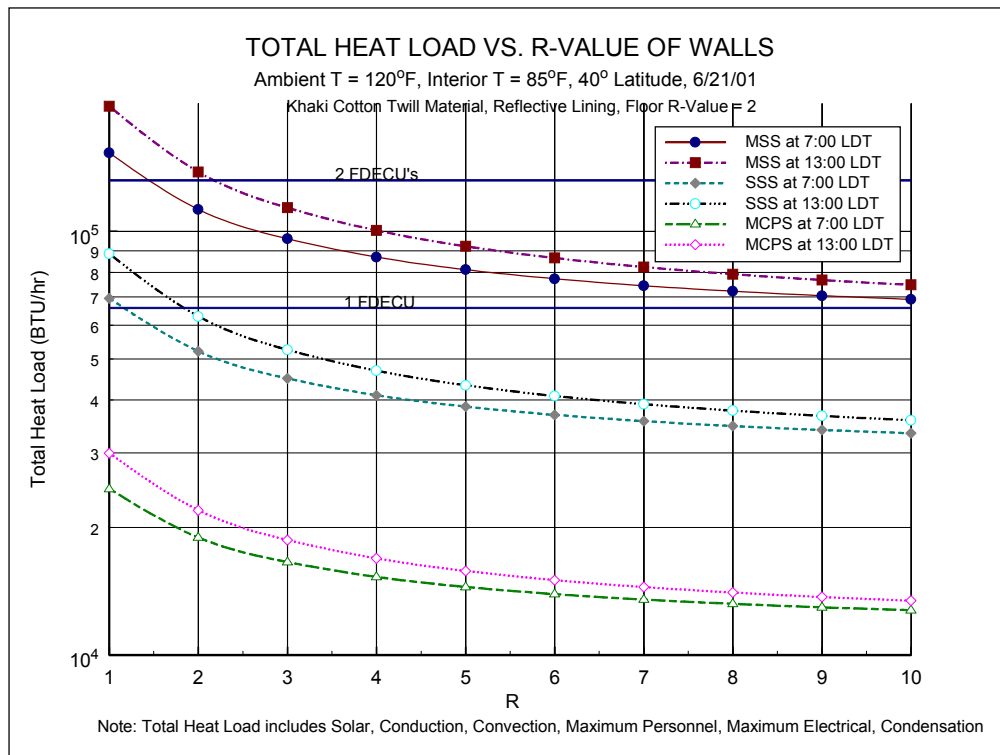
Using a maximum flow rate of  $170 \text{ m}^3/\text{hr}$ , standard atmospheric pressure of  $105 \text{ N}/\text{m}^2$ , a temperature of  $322\text{K}$ , and a temperature difference of  $20\text{K}$ , we find from (2)

#### Total Heat Load

Adding all six components, described above, together gives the total heat load on any tent or building with an inner lining. The results are shown below, for our JTCOPS example both assumed floor R-values.

$$Q_a \equiv 876000 \text{ cal/hr} = 3476 \text{ BTU/hr}$$

This is small compared to our previously calculated components and would be less for smaller air flows.



## WEIGHT ESTIMATES FOR THE HVAC EQUIPMENT AND POWER GENERATION

Although the model accepts any figure of merit for a full military specification qualified (NBC hardened) HVAC of BTU/hr/lb, a figure of merit of 100 BTU/hr/lb could be considered representative of what can be achieved today. From the thermal load requirements just established, the weight of the HVAC system can be estimated. Another figure of merit is the BTU/hr/watt (electrical). This yields the power generation requirements. When this is combined with the power generation figures of merit of watts/lb, and efficiency in watts/lb of fuel, one not only achieves the parametric dependence on the properties of the original building or tent on the weight of the HVAC and Power Generation Equipment, but moreover the total weight (and hence cost) of the (JP8) fuel consumed while operating the equipment.

## OPTIMIZATION

For the JTCOPS program, our desire was to minimize weight of the total system, and to reduce the logistics trail. If we had just accepted the thermal loads, without consideration of practical passive ways of minimizing those loads, the weight of the equipment would be a factor of 6 times higher, and the fuel consumption would be greater than 3 times for the same operating temperature and airflow within the collective protection area.

## CONCLUSIONS

What we have presented is a parametric model, ESSI COLPRO, that can be exercised with any collective protection system, albeit a building or a tent. The model can be exercised to yield the thermal loads present within a collective protection system, but moreover it yields the size and the weight of the HVAC system itself, and the associated electrical power generation requirement. It can be used to optimize the size of these major components and to reduce the logistic trail associated with the fuel consumption of such fielded systems.

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